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EFFECT OF INSULATION THICKNESS ON EVOLUTION OF PRESSURE AND TEMPERATURE IN A CRYOGENIC TANK

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ABSTRACT

A transient two-phase numerical model is developed to investigate insulation performance of cryogenic propellant tanks using finite difference method. The model includes the mathematical formulations for convective and radiative heat transfer from the ambient, heat and mass transfer across liquid-vapor using two-phase formulations. The model is validated with the experimental test data on LH2 tank. Subsequently, analyses are carried out to determine heat in-leak, evaporation rate, evolution of pressure and temperature in ullage of cryogenic tank with different foam insulation thicknesses of 20mm, 25mm, 30mm and 35mm. Analysis shows that tank pressure rise is significantly higher at lower insulation thickness. The present study brings out a correlation between rate of tank pressure rise and foam insulation thickness which shows that LN2 evaporation rate has negligible effect on tank pressure rise for simulated conditions. A minimum foam insulation thickness required for no ice formation over foam outer surface is also suggested in the study.

Keywords: *Foam Insulation, Heat In-Leak, Cryogenic, Tank Pressure, Liquid Nitrogen*

NOMENCLATRUE

T	Temperature (K)
t	Time (s)
Cp	Specific heat at constant pressure (J/KgK)
m	Mass transfer across interface (Kg/s)
K	Thermal conductivity (W/m k)
P	Tank Pressure (Pa)
Gr	Grashoff Number
Pr	Prandtl Number
Re	Reynold Number
g	Gravity (m/s2)
h	Coefficient of heat transfer (W/m2K)
Q	Heat transfer rate (W)
q	Heat flux (W/m2)
A	Surface area of heat transfer (m2)
H	Enthalpy (J/kg)
D	Tank diameter (m)
x	Ullage height (m)
X	Insulation thickness (mm)
v	velocity (m/s)
V	Volume (m^3)

Greek Symbols

μ	Dynamic viscosity (Pa-s)
σ	Stefan-Boltzmann constant ($5.67*10^{-8} W/m^2K^4$)

β	Coefficient of Thermal Expansion (K^{-1})
α	Absorptivity
ρ	Density (Kg/m^3)
ψ	Parameter for tank geometry (m^{-1})
φ	Parameter for thermal properties ($Pa\text{-}m^2/s$)

Subscripts

ull	Ullage
int	Interface
liq	Liquid
w	Wall
ins	Insulation
f	Fluid
a	Ambient gas
o	Outer insulation surface
I	Inner insulation surface

INTRODUCTION

The proper design of insulation system on a cryogenic propellant tank is vital for the nominal operation of cryogenic engine. The significant external heat load would affect the propellant thermal conditions during the mission which would affect the engine performance. Flight cryogenic tank uses foam insulation. Proper thickness of foam insulation is required to maintain the propellant temperature within allowable limits and also avoids tank pressure rise beyond a limit during the entire mission. Cryogenic engine requires minimum tank pressure at the start which can also be met by properly designing insulation thickness. Insufficient insulation can lead to possible ice formation on outer tank surface, which would increase the lift-off mass resulting payload penalty. Thus, a detailed thermal analysis of cryogenic tanks insulation system is essential to achieve nominal performance of the mission.

Several investigations were reported in literature for insulation of cryogenic tanks. Wexler [1] proposed a model which describes the evaporation rate of liquid helium in cryogenic tank. A heat transfer study in cryogenic tank was presented by Khemis et al [2]. The numerical model developed by Khemis et al [2] showed large disparity with their experimental data. Later a brief review was given by Khemis et al [3] through the experimental study. Subsequently, Boukeffa et al [4] have experimentally evaluated the heat losses in cryogenic cryostat.

A numerical heat transfer study considering convective flux over the cryogenic tank was carried out by Pictet [5]. Arsonval [6] proposed the insulation system for tank using double wall with vacuum annular section. This tank was later improved by Dewar [7] to store hydrogen liquid at 20.4 K. Several experimental studies were carried out by the researchers [8-12] to investigate the heat transfer in cryostat from ambient. Shu et al.[13] recommended for an optimal insulation of a nitrogen cryostat with a number of screen layers.

Though several numerical studies are carried out to determine heat transfer in cryogenic tank with insulation, a comprehensive numerical model for determining the effect of foam insulation system on tank thermodynamics are unavailable. Thus, the present study aims at bringing out the impact of foam insulation thickness on heat in-leak, tank pressure, fluid temperature and ice formation over the tank surface. The model uses a finite difference approach to predict thermo-fluid transients at liquid-vapor interface of cryogenic propellant tank.

FORMULATION AND SOLUTION METHODOLOGY

A transient two phase flow and thermal model is developed in SINDA/FLUINT simulator. The model is used to analyze the evolution of temperature and pressure inside the tank considering conduction, convection, boiling, condensation, and heat and mass transfer across the interface. A schematic of the propellant tank showing various heat transfer mechanism is depicted in Fig.1.

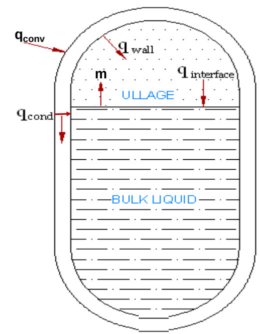

FIGURE.1 SCHEMATIC OF PROPELLANT TANK WITH DIFFERENT HEAT TRANSFER MODES

A cylindrical tank of 0.5m diameter and 1m height is taken for the analysis. The tank is foam insulated and its thickness varies as 20mm, 25mm, 30mm and 35mm which is the investigation parameter for the analysis. The tank is 80% filled with saturated LN2 at 1bar pressure. Outer boundary of the foam is provided with solar and ambient heat fluxes. Transient analyses are carried out for 15min duration, within which the tank pressure rises due to external heat loads. The model uses mathematical formulations for heat transfer from ambient to tank surface by convection and radiation and heat transfer from the ullage gas to the wall as well as that to the liquid nitrogen. Three-dimensional conduction, with temperature dependent material properties, is modeled for the tank walls and foam insulation layers. Thermodynamic state equation is used to solve for fluid conditions. Iterative analysis is carried out with various insulation foam thickness of 20mm, 25mm, 30mm and 35mm. The effect of insulation thickness on various parameters like tank

pressure, ullage temperature, heat in-leak and evaporation rate are investigated which are the key parameters used to ensure the thermal requirements for an optimum insulation thickness.

The heat transfer modes considered in the analysis are expressed mathematically as below:

Heat transfer from ullage to liquid

Heat, from ullage to bulk liquid, transfers through interface. However, the heat transfer from ullage to interface and interface to liquid are equal.

$$Q_{ull-liq} = Q_{ull-\mathrm{int}} = Q_{\mathrm{int}-liq} \tag{1}$$

Heat transfer from the interface to liquid is expressed as:

$$Q_{\mathrm{int}-liq} = h_{\mathrm{int}-liq} A_{\mathrm{int}} (T_{\mathrm{int}} - T_{liq}) \tag{2}$$

The heat transfer from the ullage gas to interface is due to convection. Since, diffusing or condensing constituent carries its own individual enthalpy, there is an additional heat transfer across interface due to mass transfer of these species. Effective heat transfer from the ullage gas to interface is expressed as:

$$Q_{ull-\mathrm{int}} = h_{ull-\mathrm{int}} A_{\mathrm{int}} (T_{ull} - T_{\mathrm{int}}) + mH \tag{3}$$

m and H are mass and enthalpy of nitrogen transfer across interface. m is positive for condensation and negative for vaporization of the liquid and can be expressed as

$$m = \frac{[h_{\mathrm{int}-liq} A_{\mathrm{int}} (T_{\mathrm{int}} - T_{liq}) - h_{ull-\mathrm{int}} A_{\mathrm{int}} (T_{ull} - T_{\mathrm{int}})]}{H} \tag{4}$$

It has been assumed that the heat transfer is due to natural convection with the heat transfer coefficient being expressed by the empirical correlation [14]:

$$h = K_H C \frac{k_f}{l_s} X^n \tag{5}$$

Where:

$$X = (Gr)(\mathrm{Pr}) \tag{6}$$

and

$$Gr = \left(\frac{l_s^3 \rho_f^2 g \beta_f |\Delta T|}{\mu_f^2} \right) \tag{7}$$

$$\mathrm{Pr} = \left(\frac{c_{pf} \mu_f}{K_f} \right) \tag{8}$$

Constant (C) is 0.54 and 0.27 for heat transfer from ullage to interface and interface to liquid respectively. n = 0.25 and K_H (heat transfer adjustment factor) is set to 1.0. The length scale (l_s) is set to the diameter of the tank.

Heat transfer from wall to Ullage

The heat transfer from wall to ullage gas is expressed as:

$$Q_{wall} = h A_{ull-w} (T_w - T_{ull}) \tag{9}$$

It has also been assumed that the heat transfer is due to natural convection and heat transfer coefficient is expressed as:

$$h = 0.508 \frac{k_f}{l_s} \mathrm{Pr}^{0.5} \left(0.952 + \mathrm{Pr}\right)^{-0.25} Gr^{0.25} \tag{10}$$

k_f is thermal conductivity of fluid and l_s is height of tank at particular point where h is calculated.

Heat transfer from ambient

Convection and radiation models are used to calculate incident heat on the tank. Coefficient of convective heat transfer on tank surface is calculated by using Churchill and Bernstein [15] formulation based on wind velocity of 2m/s. Heat transfer through radiation is calculated by using solar heat flux data (q_{solar}) and altitude dependent ambient temperature value. Total heat transfer from outside is express as:

$$Q_{total} = Q_{conv} + Q_{rad} \tag{11}$$

$$Q_{conv} = h_{conv} A_{\tan k} \left(T_{outer} - T_{\tan k} \right) \tag{12}$$

$$h_{conv} = \left(0.3 + \frac{0.62 \,\mathrm{Re}^{\,0.5} \,\mathrm{Pr}^{0.33}}{[1 + (0.4/\mathrm{Pr})^{2/3}]^{0.25}} \left[1 + \left(\frac{\mathrm{Re}_D}{28200} \right)^{5/8} \right]^{4/5} \right) \frac{K_a}{D} \tag{13}$$

$$\mathrm{Re} = \frac{\rho . v_a . D}{\mu_a} \tag{14}$$

$$Q_{rad} = q_{solar} A_{\tan k} + \sigma \alpha A_{\tan k} \left(T_{outer}^4 - T_{\tan k}^4 \right) \tag{15}$$

To calculate heat-in-leak from ambient to the tank wall, three dimensional transient heat conduction equations are solved considering the variation of thermal conductivity of the insulation with temperature. Three-dimensional transient heat conduction equation is given by following expression considering K as a function of temperature:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = \frac{\rho C_P}{K(T)} . \frac{\partial T}{\partial t} \tag{16}$$

All the above mentioned heat transfer modes represent the thermal mathematical model which have been incorporated in the analyses.

RESULTS AND DISCUSSION

Transient two-phase thermal analysis is carried out to determine the effectiveness of foam insulation for various thermodynamic parameters such as tank pressure, ullage temperature, evaporation rate and heat in-leak. The model is first validated using LH2 tank pressure data from launch vehicle. Subsequently, the mathematical model is applied to study the effect of foam thickness on tank thermodynamic parameters and foam surface icing. Results of the analysis are discussed below.

Mathematical Model Validation

The numerical model is validated with the experimental data obtained from the test on LH2 tank. Corresponding tank dimensions are considered for validation. Figure 2 shows the comparison of the numerical model with the experimental test data of LH2 tank pressure during ground hold condition.

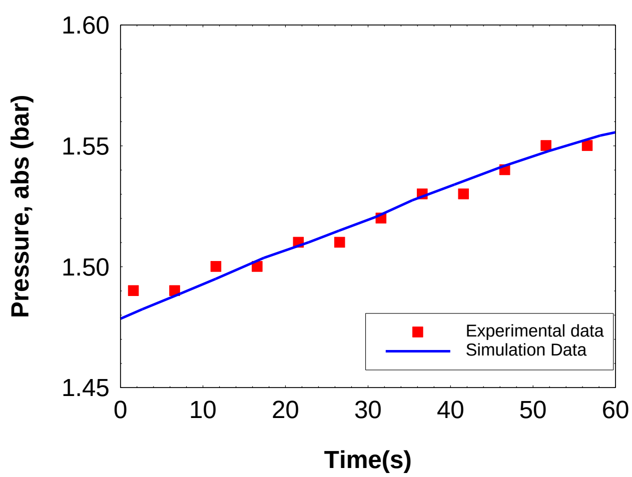

FIGURE.2. MODEL VALIDATION WITH EXPERIMENT DATA

Steady pressure rise of 0.7 bar is measured in 60s during test. It can be seen in Fig.2. that the pressure rise from the numerical model is well corroborated with the measured test data. The mathematical model is further used for insulation effectiveness studies as discussed below.

Effect of insulation thickness on tank pressure

Tank pressure evolution: Tank pressure evolution during the 900s transient analysis for various foam thicknesses are shown in Fig.3.

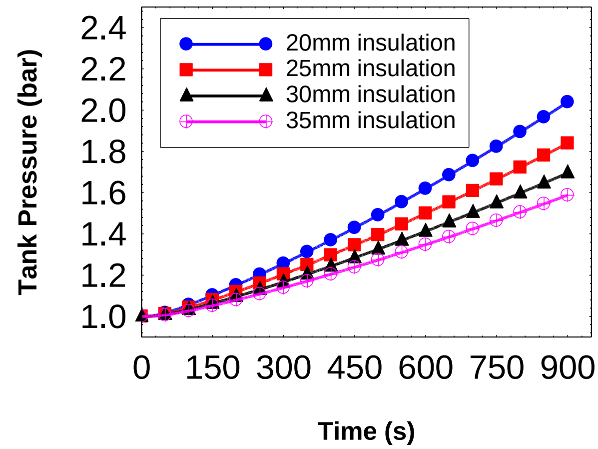

FIGURE 3.TANK PRESSURE WITH DIFFERENT INSULATION THICKNESS

It can be seen from the figure that pressure rise is maximum for 20mm insulated tank and reduces with thicker tank insulation. This is because the tank pressure depends largely on ullage temperature which increases due to external heat sources. Poor insulation allows more heat into the tank leading to higher pressure rise. Pressure rise beyond tank structural limit is fatal for cryogenic storage tanks. Hence, better insulation is essential for safe operation of cryogenic tanks.

Rate of pressure rise: Figure.3 shows maximum tank pressure rise for 20mm insulation thickness. From the figure, it is also evident that the rate of pressure rise is different for each insulation thicknesses. Since rate of pressure rise is steady close to 900s, a study is carried out to find a correlation between the rate of tank pressure rise and insulation thickness.

Pressure rise in cryogenic tank is due to temperature rise as well as liquid evaporation. LN2 evaporation has two effects on tank pressure; raising it through ullage mass addition and dropping it by cooling the ullage. In the following mathematical formulation, pressure rise due to evaporation is assumed negligible and ambient heat sources is assumed to affect the pressure rise significantly, which gets convected into the ullage through conductive heat transfer across insulation thickness. Pressure rise rate in ullage based on temperature rise rate can be expressed as:

$$\frac{(dP)_{ull}}{dt} = \rho R \frac{(dT)_{ull}}{dt} \tag{17}$$

The heat balance across the foam can be written as:

$$Q_{amb} = mCp\frac{(dT)_{ins}}{dt} + k_{ins}A_{w-ull}\frac{(T_o - T_i)}{X} \quad (18)$$

$$k_{ins}A_{w-ull}\frac{(T_o - T_i)}{X} = h_{w-ull}A_{w-ull}(T_w - T_{ull}) \quad (19)$$

$$h_{w-ull}A_{w-ull}(T_w - T_{ull}) = m_{ull}Cp_{ull}\frac{(dT)_{ull}}{dt} \quad (20)$$

where, X is the insulation thickness. Since heat transfer path for ullage temperature rise is conductive through insulation, rate of pressure rise can be parameterized from Eqns. 17, 18, 19 and 20 as follows:

$$\frac{(dP)_{ull}}{dt} = \left(\frac{k_{ins}R(T_o - T_i)}{Cp}\right).\left(\frac{A_{w-ull}}{V_{ull}}\right).\frac{1}{X} \quad (21)$$

$$\frac{(dP)_{ull}}{dt} = (\phi).(\psi).\frac{1}{X} \quad (22)$$

Where

$$\phi = \frac{k_{ins}R(T_o - T_i)}{Cp}, \quad (23)$$

$$\psi = \frac{A_{w-ull}}{V_{ull}} \quad (24)$$

Φ is thermal dependent parameter assuming steady pressure rise with $(T_o\text{-}T_i)$ constant, whereas ψ is tank dimension dependent parameter.

$$\psi = \frac{1}{x} + \frac{4}{D} \qquad \text{For cylindrical ullage}$$

$$\psi = \frac{(D/x)}{[(D/2)-(x/3)]} \qquad \text{For spherical ullage}$$

Where, x is tank ullage height and D is the tank diameter.

Thus, pressure rise parametric equation independent of tank dimension can be written from Eqn. 22 as

$$\frac{(dP)_{ull}}{dt}.\frac{1}{\psi} = [\phi].\frac{1}{X} \quad (25)$$

From the Eqn 25, it is evident that pressure rise rate follows inverse power trend with insulation thickness. For the simulation conditions, ψ value is found to be 173.56.

Subsequently, the pressure rise parameter in Eqn 25 is analysed for different insulation thicknesses of 20mm, 25mm, 30mm and 35mm, considering LN2 evaporation in the model, and results are shown in Fig. 4.

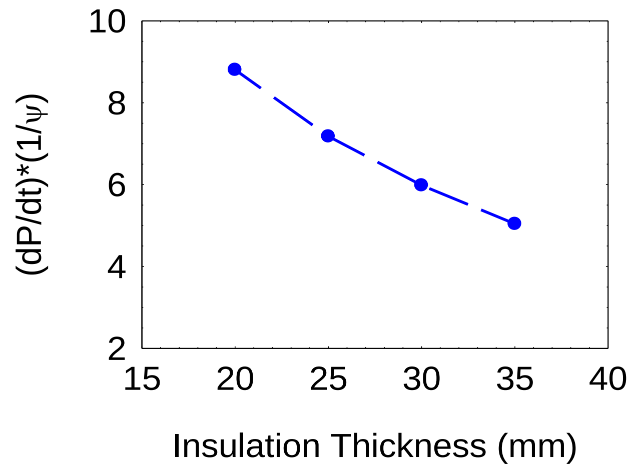

FIGURE 4.TANK PRESSURE RISE RATE WITH DIFFERENT INSULATION THICKNESS

The analysis results match within ±3% of the correlation for rate of pressure rise given in Eq. 25, which was formulated assuming negligible LN2 evaporation rate. Variation in mathematical correlation and simulation results could be attributed to evaporation been considered in simulation. This shows that effect of rate of evaporation of LN2 on tank pressure rise for the simulation conditions is negligible and pressure rise in the tank is mainly due to heating of ullage by external sources.

The inverse power correlation stated in Eq. 25 shows that higher insulation thickness has lesser effect in rate of pressure rise whereas reducing the thickness of insulation, significantly increases the rate of pressure rise, which is undesirable.

Effect of insulation thickness on evaporation rate

Different foam thicknesses are taken to study its effect on rate of LN2 evaporation during the transient analysis. Variation of LN2 mass evaporation rate, per liquid surface area, for different insulation thicknesses is shown in Fig.5. LN2 upon heated up from external sources gains latent heat energy which leads to evaporation. From the plot, it is evident that liquid evaporation rate reduces as tank insulation thickness increases. In cryogenic propellant tanks, evaporation of propellant results in reduction in useful propellant mass thereby, reducing engine thrust

duration. Hence, it is vital to reduce liquid evaporation in cryogenic tanks by increasing the insulation thickness.

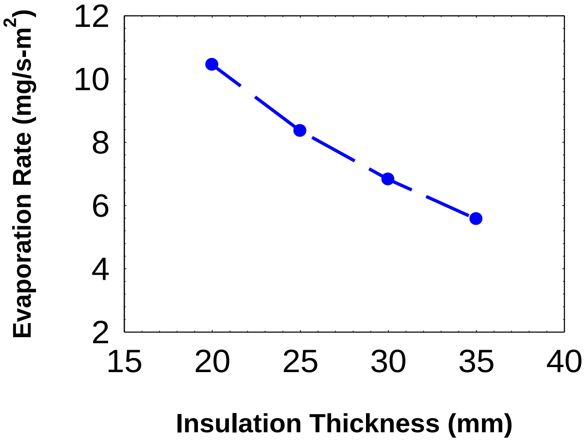

FIGURE 5. EVAPORATION RATE WITH DIFFERENT INSULATION THICKNESS

Effect of insulation thickness on ullage temperature

The study of evolution of ullage temperature for 20mm, 25mm, 30mm and 35mm insulation thicknesses on the tank is carried out and results are presented in Fig.6.

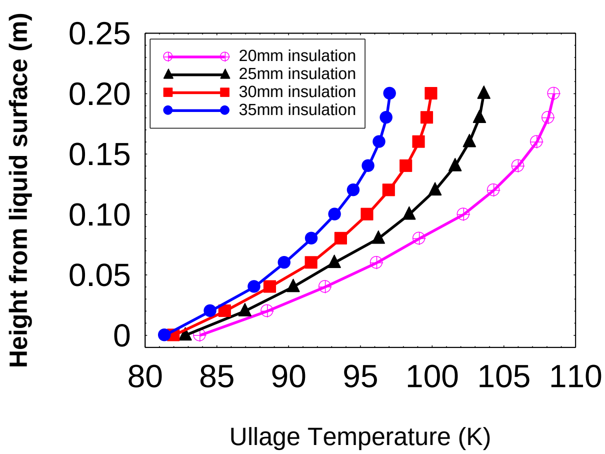

FIGURE.6 GAS TEMPERATURE IN ULLAGE WITH DIFFERENT INSULATION THICKNESS

It is to be noted that the graph is plotted from liquid-ullage interface to the top of the tank. The interface is always at saturation temperature and gradient in ullage temperature starts afterwards as seen in Fig.6. As the insulation thickness reduces, maximum value of ullage temperature increases as seen in Fig.6. Liquid surface temperature also increases which is equivalent with saturation temperature rise corresponding to tank pressure increment.

Effect of insulation thickness on foam temperature

The analysis is carried out to study the foam insulation temperature profile, at 0.1m from top of the tank which is at ullage side, for different insulation thicknesses. Variation in foam temperature pattern for different insulation thicknesses at 900s are shown in Fig.7.

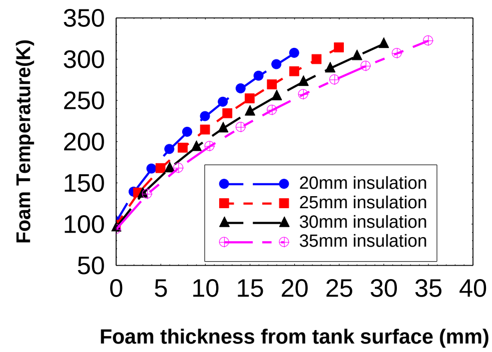

FIGURE 7.FOAM TEMPERATURE ALONG THE THICKNESS

It is evident from the figure that higher insulation thickness results in higher temperature at outer surface of the foam insulation. This shows that the ambient heat-in leak will be minimum for higher insulation thicknesses as convective heat transfer from ambient is directly proportional to temperature difference between foam outer surface and ambient. In another perspective, temperature gradient within the foam is highest for 20mm insulation which means conductive heat transfer through the foam is highest, resulting in maximum heat in-leak.

Effect of insulation thickness on external heat in-leak

Heat in-leak: Analysis is carried out to study heat in leak variation with different insulation thicknesses and results are shown in Fig.8.

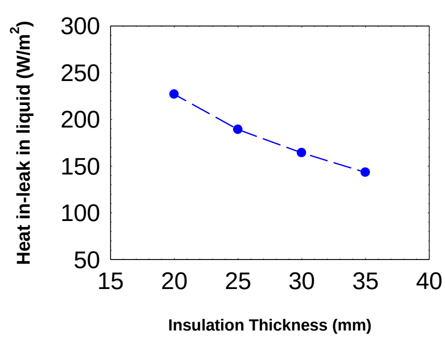

FIGURE 8. HEAT-IN-LEAK INSIDE THE TANK WITH DIFFERENT INSULATION THICKNESS

The results show that heat in-leak reduces with increase in insulation thickness. This result is expected as foam insulation temperature profile shown in Fig.7. denotes higher temperature at foam surface for higher insulation thickness, resulting in low ambient convective heat transfer, as discussed in previous section.

Ice formation on tank surface

Water vapor present in ambient condenses and freezes over the tank insulation outer surface, when temperature at the outer surface of the insulation drops below 273K. This results in deposition of ice layer over the foam surface, and may peel off during ascend phase of launch vehicle. Icing phenomenon leads to undesirable increment of lift-off mass and its peel-off may result in damage of surrounding hardware. Analysis is carried out to determine the foam outer surface temperature, at 0.4m from tank base which is at liquid side, for different foam thicknesses. Solar heat flux is considered in one case and not considered in other to analyze worst case scenario. Foam outer surface temperature variation with foam insulation thickness is shown in Fig.9.

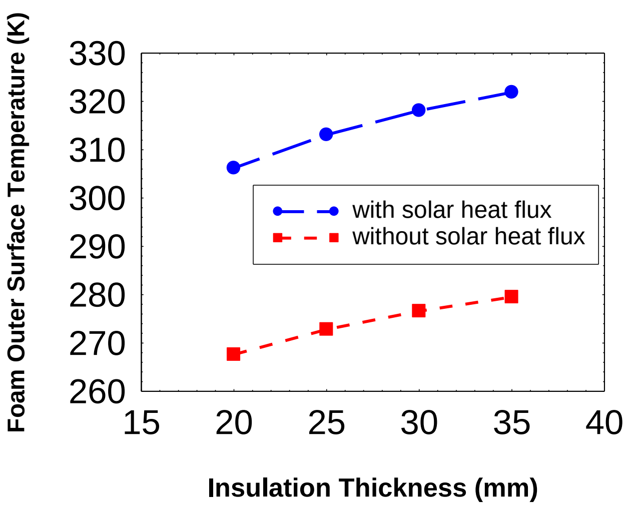

FIGURE.9 FOAM OUTER SURFACE TEMPERATURE WITH DIFFERENT INSULATION THICKNESS

It is evident from the figure that icing will not be seen when solar heat flux is incident on the tank. But, in case the tank is place in shade where little or no solar heat flux is incident, possibility of ice formation over foam surface increases as foam insulation thickness reduces. Lower the insulation thickness, lower will be the foam outer surface temperature which would raise the ambient heat in-leak. Figure 9 shows that for thickness of insulation less than 30mm, foam surface temperature is lower than 273K and thus, ice will be formed over the surface. Hence, insulation thickness of 30mm or above is required to avoid icing over foam insulation.

CONCLUSION

Transient two-phase thermodynamic model of cryogenic LN2 tank is developed to investigate the effect of foam insulation thickness on tank parameters such as pressure, ullage temperature, heat in-leak and liquid evaporation rate. Validation of mathematical model is carried out with experimental test data on LH2 tank. Subsequent transient analysis shows that increasing the insulation thickness reduces the heat in-leak and thereby decreases tank pressure rise which is crucial for flight cryogenic tanks. Ullage temperature and liquid evaporation rate are found to be lower when higher insulation thicknesses are used. Pressure rise rate for different insulation thicknesses are mathematically analysed to bring out a correlation independent of tank geometry. The mathematical correlation when compared with analysis results shows that rate of LN2 evaporation has negligible effect in rate of pressure rise for the simulation conditions considered. The correlation also shows that for lower insulation thickness, rate of pressure rise is significantly higher. Effect of presence and absence of solar heat flux incident on the tank is also determined for different insulation thicknesses. When little or no solar heat flux is incident on the tank, foam outer surface temperature drops below 273K for insulation thicknesses less than 30mm, leading to formation of ice over the foam surface. To avoid its adverse effect, insulation thicknesses of 30mm or above are recommended.

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